



Experimental study on heat transfer of oscillating flow of a tubular Stirling engine heater



Gang Xiao, Conghui Chen, Bingwei Shi, Kefa Cen, Mingjiang Ni*

State Key Laboratory of Clean Energy Utilization, Zhejiang University, 38 Zheda Road, Hangzhou 310027, China

ARTICLE INFO

Article history:

Received 17 September 2013

Received in revised form 25 November 2013

Accepted 3 December 2013

Available online 20 December 2013

Keywords:

Stirling engine
Oscillating flow
Tubular heater
Heat transfer

ABSTRACT

Stirling engine heaters are characterized by oscillating flows which affect heat transfer coefficients greatly. A 36-tube Stirling engine heater with a piston-link drive machine is used to study heat transfer characteristics of oscillating flows. The influences of the overall heating power, oscillating frequency and gas pressure on the heat transfer characteristics are investigated. To raise the working gas pressure is positive to decrease the wall temperature and to improve the heat transfer. When the pressure varies from 0.1 to 0.4 MPa, the wall temperature reduces 17 °C and the heat input increases 10 W. The time-averaged heat transfer coefficients reach the maximum value of 78.0 W/(m² K) among the testing conditions when the working gas pressure is 0.4 MPa and the revolution is 420 rpm. An oscillating heat transfer correlation is derived based on the experimental data which are obtained under situations close to an actual Stirling engine's flow region, where Re and Re_{ω} are in the ranges of 740–4110 and 12–71, respectively. The estimated uncertainty for the heat transfer coefficient is usually within 7.69%, and the determination coefficient of a regression fitted correlation is 0.97. The proposed correlation is respected to predict the heat transfer coefficients of oscillating flows for practical design of tubular heaters, while the classical unidirectional steady correlations are not suitable, especially at high Re conditions.

© 2013 Elsevier Ltd. All rights reserved.

1. Introduction

Stirling engine heaters are characterized by oscillating flows which affect the heat transfer coefficients greatly. A lot of studies on the flow and heat transfer characteristics of oscillating flows have been done to get a deep understanding of performance of Stirling engines.

Simon and Seume [1] summarized the features of oscillating flows and heat transfer in Stirling engines, and found out that many modern Stirling engines were operated in or near the laminar to turbulent transition region. Table 1 lists some classical laminar and turbulent steady unidirectional correlations which are used to get approximations of convective heat transfer coefficient in Stirling engines. Urieli and Berchowitz [2] introduced a so-called Quasi steady flow model, as the Eq. (d), assuming that there was a steady flow condition in each integration time. Kuosa et al. [3] compared four laminar oscillating correlations, i.e. Eq. (e–h), and indicated that some of the laminar oscillating correlations gave almost the same Nusselt number to turbulent unidirectional equations. However, many experimental researches showed different results. Shin and Nishio [4] adopted the Womersley number as a dimensionless value of oscillating frequency to consider the viscous effects, and derived a heat transfer correlation, as the Eq. (i),

of oscillating flows in tubes using a numerical study under an assumption of isothermal and extremely thin walls. Fan et al. [5] employed this correlation to predict heat transfer coefficient of a long tube of 2.1 mm inside diameter and 0.41 mm wall thickness. His experiments showed the correlation gave reasonable results at low oscillating frequencies and provided under-predicted values at higher frequencies. Kanzaka and Iwabuchi [6,7] focused on the influence of piston phase difference on the heat transfer coefficients in a singular tube, proposed a correlation considering the piston phase difference, as the Eq. (j), which was verified later by the experiments using a Stirling engine. Bouvier et al. [8] experimentally investigated oscillating flows in a circular tube of 5.7 m long, 48.3 mm outer diameter and 3.2 mm thickness, and found that temperature oscillations in the wall could be ignored. Akdaga and Ozguc [9] studied the oscillating flow of water in a vertical annular liquid column with a constant heat flux, and obtained an empirical correlation, i.e. the Eq. (k), for a cycle averaged Nusselt number as a function of kinetic Reynolds number and dimensionless amplitude. These studies indicated that oscillating flow factors have great impacts on heat transfer performance, including kinetic Reynolds numbers, frequencies and fluid displacements. While there is no widely recognized conclusion, especially in correlations for heat transfer of oscillating flows. Besides, most of the above experimental studies were carried out using singularly slender tubes, not real Stirling engine heaters which always had tens of tubes.

* Corresponding author. Tel.: +86 057187951369.

E-mail address: ceu-ni@zju.edu.cn (M. Ni).

Nomenclature

A	heat transfer area, m^2
A_0, A_w, A_R	dimensionless fluid displacement
c_p	working gas specific heat capacity at constant pressure, $\text{J}/(\text{kg K})$
c_t	correction factor for gas
C	correction term
d	tube diameter, m
f	Darcy resistance coefficient
f_r	Reynolds friction factor
g	gravitational acceleration, N/kg
Gr	the Grashof number
h	heat transfer coefficient, $\text{W}/(\text{m}^2 \text{K})$
H	characteristic length, m
I	current, A
L	overall length of a tube, m
n	number of the heater tubes
Nr	revolutions per minute of the flywheel, rpm
Nu	the Nusselt number
P	pressure, MPa
Pr	the Prandtl number
R^2	determination coefficient of the regression fitted correlation
Re	the Reynolds number
Re_{max}	the maximum Reynolds number
Re_ω	the kinetic Reynolds number
T	temperature, K
u	working gas velocity based on mean piston velocity, m/s
U	voltage, V
w	uncertainty
W_0	the Womersley number
x_{max}	fluid displacement in heater tube, m

Greek symbols

α	coefficient of cubical expansion, $1/\text{K}$
ε	emissivity of material
λ	conductivity of material, $\text{W}/(\text{m K})$
Λ	dimensionless fluid displacement
μ	viscosity of fluid, $\text{N s}/\text{m}^2$
ν	kinetic viscosity of fluid, m^2/s
φ	heat, W
ω	oscillating frequency, rad/s

Subscripts

b	buffer
c	cooler
C	convection
cal	calculated results by formerly proposed correlations
e	effective
exp	experimental results
g	working gas
h	heater
i	inner wall
ins	insulations
o	outer wall
obj	stands for insulations or heating medium
oil	heating medium
p	displace piston
r	regenerator
R	radiation
sur	the surrounding
w	tube wall

This paper investigated the heat transfer characteristics of oscillating flows of a 36-tube Stirling engine heater, whose operating conditions were close to the flow regions of real engines, and derived an empirical oscillating heat transfer correlation to predict oscillating flow heat transfer performance for promoting practical design of tubular heaters.

2. Experimental setup and methods

An experimental apparatus was constructed to study the heat transfer characteristics of oscillating flows in a tubular heater. A schematic of the apparatus is shown in Fig. 1. The apparatus consists of a device body, an oil-bath heating block to heat the tubular heater, a heat sink via cooling water, a gas supplement unit, and a piston-link drive machine with a crosshead. The device body has three heat exchangers, i.e. a heater, a regenerator and a cooler, and the structure is modeled after displacement type Stirling engines. A motor connected to a crankshaft drives the displacer, which makes working gas flow cyclically between the heater and the cooler through the regenerator.

The heater is made of an array of 36 U-shaped stainless steel tubes. These tubes form a sort of cage around and above the cylinder of a 110 mm diameter. The outer diameter and overall length of each tube are 6 mm and 298 mm, respectively. The annular wire mesh regenerator is set up in a stainless steel holder. The cooler contains 30 aluminium tubes with fins, whose inside diameter and length are 5 mm and 70 mm, respectively. The volumes of the heater, regenerator, cooler and displacer sweeping are listed in Table 2.

At the start of each experiment, the seals were checked. The sealing units on the displacer contain three piston rings and a

guide ring made of Rulon, preventing gas short-circulating the heat exchangers. Stepseal in combination with Variseal were chosen as piston rod seals to seal-off the pressurized working gas within the cylinder. O-rings were applied at other matching parts as static seals. Pressures in the cold end of cylinder and in the buffer were measured to test sealing effects of rod seals. A required initial pressure (0–0.8 MPa) in cylinder was controlled by the gas supplement unit.

Nitrogen was applied as the working gas, and the revolution varied between 150 and 500 rpm during all experiments. The temperature of dimethicone was between 70 and 250 °C in the heat transfer tests. The cooling water rate was kept around 40 L/h. Testing data was recorded by a computerized data acquisition system, with measuring points and locations illustrated in Table 3.

The mass of working gas fluctuates as gas travels through different temperature sections in a cycle. The reductions of the mass of working gas in Fig. 2 show the leakage behaviors at different revolutions. It is interesting to observe that higher oscillating frequency helps to suppress gas leakage. This is mainly because the seals expand due to friction generates more heat at higher frequency. Fig. 3 shows the static leakage behaviors at different buffer pressures. It is found that the smaller pressure difference between the cylinder and the buffer, the better sealing.

Dimethicone was used as the heating medium, i.e. an oil bath, which was heated by an electric ribbon heater and surrounded by thermal insulations. In heat transfer tests, the overall heating powers were maintained at a steady state and were evaluated by the voltage and current values. The temperatures of the outer wall of insulation, the heating medium, and the surroundings, were measured by seven thermocouples. The effective heat transferred

Table 1
Heat transfer correlations in literatures.

Item	Expression	Application range	Author	Remarks
(a)	$Nu = 0.023 Re^{0.8} Pr^{0.4} (T_w/T_g)^{0.5}$	$10^4 < Re < 1.2 \times 10^5$; $0.7 \leq Pr \leq 120$; $L/d \geq 60$	Dittus–Boelter correlation [10]	Turbulent steady flow
(b)	$Nu = \frac{(f/8)(Re-1000)Pr}{1+12.7\sqrt{f/8}Pr^{1/3}} [1 + (d/L)^{2/3}] C_t$	For gas: $C_t = (T_g/T_w)^{0.45}$; $T_g/T_w = 0.5 \sim 1.5$; $f = (1.82 \lg Re - 1.64)^{-2}$; $2300 \leq Re \leq 10^6$, $0.6 \leq Pr \leq 10^5$ $0.48 \leq Pr \leq 16700$; $0.0044 \leq \mu_g/\mu_w \leq 9.75$; $[RePr/(L/d)]^{1/2} (\mu_g/\mu_w)^{0.14} \geq 2$ $Re < 2000$; $f_r = 16$; $2000 < Re < 4000$; $f_r = 7.343 \times 10^{-4} Re^{1.3142}$; $Re > 4000$; $f_r = 0.0791 Re^{0.75}$	Gnielinski correlation [10]	Turbulent steady flow
(c)	$Nu = 1.86 Re Pr / (L/d)^{1/3} (\mu_g/\mu_w)^{0.14}$	$0.0044 \leq \mu_g/\mu_w \leq 9.75$; $[RePr/(L/d)]^{1/2} (\mu_g/\mu_w)^{0.14} \geq 2$ $Re < 2000$; $f_r = 16$;	Sieder–Tate correlation [10]	Laminar steady flow
(d)	$h = f \mu C_p / (2 d Pr)$	$2000 < Re < 4000$; $f_r = 7.343 \times 10^{-4} Re^{1.3142}$; $Re > 4000$; $f_r = 0.0791 Re^{0.75}$	Urieli and Berchowitz [2]	Steady flow
(e)	$Nu = -0.494 + 0.0777 [A_{co}/(1 + A_{co})]^2 Re^{0.7} - 0.00162 Re^{0.8} Re_{co}^{0.8}$	$A_{co} = (d/L)(Re/Re_{co})$; $7 < Re < 7000$; $7 < Re_{co} < 180$; $0.06 < A_{co} < 2.21$	Tang and Cheng [11]	Experimental study on a periodically reversing tube flow of air
(f)	$Nu = -0.494 + 0.0777 [A_{co}/(1 + A_{co})]^2 Re^{0.7} - 0.00162 Re^{0.8} Re_{co}^{0.8}$	$A_{co} = (d/L)(Re/Re_{co})$; $11 < Re_{max} < 10995$; $1.75 < Re_{co} < 45$	Monte [12]	Analytical study of heat transfer in Stirling engines base on Eq. (e)
(g)	$Nu = 0.024 Re_{co}^{0.85} Re_{co}^{0.58}$	$A_{co} = x_{max}/d$; $0 < Re_{co} < 500$; $A_{co} = 8.5$, 15.3 , 20.4 , 34.9	Zhao and Cheng [13]	Experimental study on laminar reversing flow of air in a uniformly heated long tube
(h)	$Nu = 5.78 + 0.00918 Re_{co}^{0.969} \Lambda^{-0.367} + 0.178 (Re/Re_{max})^{0.951} \Lambda^{-0.703} Re_{co}^{0.526}$	$Re_{max} \leq 400 \sqrt{Re_{co}}$; $\Lambda = (L/2d)(Re_{co}/Re_{max})$; $50 \leq Re_{co} \leq 900$; $0.2 \leq \Lambda \leq 900$	Walther et al. [14]	Theoretical analysis of laminar oscillating tube flow
(i)	$Nu = 3.3 W_0^{0.2} + \frac{0.025 (x_{max}/L) W_0^{2/3} Pr^{1/3}}{1 + 0.016 (x_{max}/L) W_0^{2/3} Pr^{1/3}}$	$W_0 = \frac{d}{2} \sqrt{\frac{Re}{Pr}}$; $W_0 > 1$	Shin and Nishio [4]	Numerical simulation
(j)	$Nu = 0.021 Re^{0.8} Pr^{0.4} (T_w/T_g)^{0.5} C$	Correction term: $C = 0.923 + 0.750 (T_w/1000)$	Kanzaka and Iwabuchi [6,7]	Experiment study on piston phase difference influence and analysis with Schmidt cycle model based on Eq. (a)
(k)	$Nu = 1.32 A_0^{0.85} Re_{co}^{0.248}$	$1000 < Re_{co} < 4000$; $7.73 < A_0 < 12.88$;	Akdag and Ozguc [9]	Experiment study on vertical annular oscillating flow of water

into the working gas was determined by the overall heating power minus heat dissipations, which is given by [10],

$$\varphi_e = UI - (\varphi_{ins,C} + \varphi_{ins,R}) - (\varphi_{oil,C} + \varphi_{oil,R}) \quad (1)$$

$$\varphi_{obj,C} = h_{obj} A_{obj} (T_{obj} - T_{sur}) \quad (2)$$

$$\varphi_{obj,R} = 5.67 \varepsilon_{obj} A_{obj} [(T_{obj}/100)^4 - (T_{sur}/100)^4] \quad (3)$$

$$h_{obj} = (Nu_{obj} \lambda_{obj}) / H_{obj} \quad (4)$$

$$Nu_{obj} = 0.59 (Gr_{obj} Pr_{obj})^{1/4} \quad (5)$$

$$Gr_{obj} = g \alpha_{obj} H_{obj}^3 (T_{obj} - T_{sur}) / \nu_{obj}^2, \quad 1.43 \times 10^4 \leq Gr \leq 3 \times 10^9. \quad (6)$$

The heater tubes immersed in the heating medium were instrumented with thermocouples near the ends. The temperature distribution was assumed fairly uniform. Besides, the hypothesis of neglecting temperature oscillating in the wall is acceptable in this certain operating conditions [8]. Thus the average temperature of outer tube wall was detected and the average temperature of inner wall could be calculated according to the physical thermal conductivity properties of the heater. Thermocouples were inserted into connections between the hot end of cylinder and the heater and between the heater and the regenerator, respectively, to get the average gas temperatures in the heater tubes. The time-averaged heat transfer coefficient was given by,

$$T_{w,i} = T_{w,o} - \frac{\varphi_e (d_o - d_i)}{2 \lambda \pi d_o L} \quad (7)$$

$$h_g = \varphi_e / \pi d_i L (T_{w,i} - T_g) \quad (8)$$

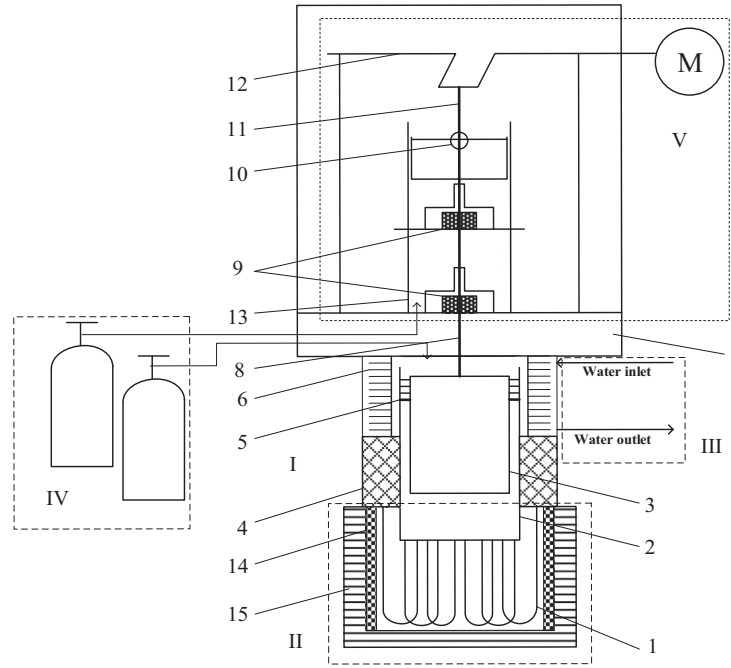
3. Results and discussions

3.1. Heat transfer experiments of The heater

Fig. 4 details the heat transfer performance at various overall heating powers. Apparently, a higher overall heating power yields higher wall temperatures of the heater tubes. The effective heat input from the heat block to the working gas increases with the overall heating power at first and then drops to some extent at the end when other operating conditions keep constant. Fig. 5 shows that the temperatures of the wall and the heating medium decrease when the working gas pressure raises and the overall heating power remains unchanged. Fig. 5 also indicates that raising the working gas pressure is positive to improve the effective heat input. When the pressure varies from 0.1 to 0.4 MPa, the wall temperature reduces 17 °C and the heat input increases 10 W.

Figs. 6 and 7 present relations between the heat transfer coefficient, the oscillating frequency, and the gas pressure. The time-averaged heat transfer coefficient is between 39.9–78.0 W/(m² K) during the tests. A higher frequency brings the oscillating flow with a larger core and a thinner boundary layer, which enhance the heat transfer. Fan et al. [5] reported that these characteristics could enhance the heat transfer both between the inner tube wall and the working gas and along the axial direction of the fluid. An improvement on heat transfer by adding the pressure in cylinder indicates that large mass flow rate is also beneficial.

The accuracy of the volt-ammeter was taken as 1% ±2 divisions, and that of the K-type thermocouples could be 0.1 °C within the experiment range. The uncertainty analysis of the time-averaged heat transfer rate was performed with the main source of errors [9,15] as follows.



1-heater tubes, 2-cylinder, 3-displacer, 4-regenerator, 5-piston ring, 6-cooler, 7-frame plate, 8-piston rod, 9-piston rod seals, 10-crosshead, 11-connecting rod, 12-crankshaft, 13-buffer, 14- ribbon heater, 15-thermal insulations
I- device body; II- oil bath device; III-cooling water system; IV- gas supplement unit; V-drive mechanism

Fig. 1. Schematic of experimental apparatus.

Table 2
Volume of each component.

Item	Volume/cm ³
Heater	70.76
Regenerator holder	202.32
Regenerator matrix	30.78
Cooler	42.23
Displacer sweep volume	134.30

Table 3
Measuring items and location of measuring points.

Object	Location	Symbol	Measuring instruments
Pressure	Cold end of cylinder	P_c	Pressure transducer
Temperature	Buffer	P_b	Pressure gauge
	Hot end of cylinder	T_h	K-type thermocouple
	Interface between heater and regenerator	$T_{h,r}$	
	Interface between cooler and regenerator	$T_{c,r}$	
	Cold end of cylinder	T_c	
	Outer wall of heater tubes	$T_{w,o}$	
	Immersed in dimethicone	T_{oil}	
Revolution	Attached to the outer layer of insulation	$T_{ins,o}$	
	Atmosphere	T_{sur}	
	Flywheel	Nr	Tachometer
Overall heating power	Ribbon heater	U, I	Ampere-voltage meter

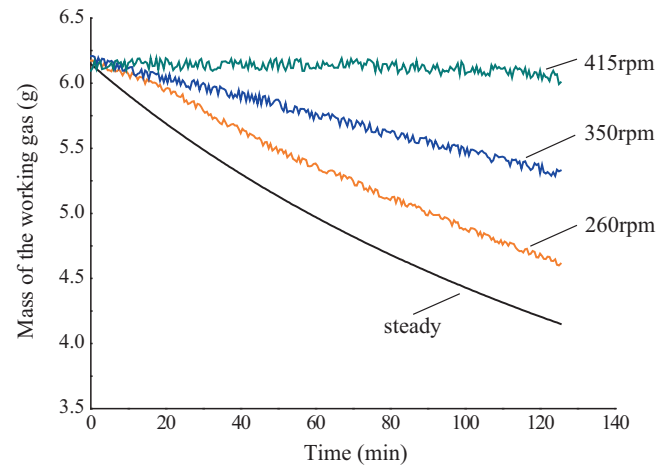


Fig. 2. Leakage behaviors at different revolutions.

$$w_{hg} = \sqrt{(\partial h_g / \partial \varphi_e)^2 w_{\varphi_e}^2 + (\partial h_g / \partial T_{w,i})^2 w_{T_{w,i}}^2 + (\partial h_g / \partial T_g)^2 w_{T_g}^2} \quad (9)$$

The deviation of the calculated effective heat input is less than 12.5 W. The estimated uncertainty for the heat transfer coefficient is 5.42–7.69%. Experimental results are listed in Table 4.

For further investigations, test data was converted into dimensionless parameters. The gas flow inside the test section with a constant overall volume is driven by the displacer. Assumptions

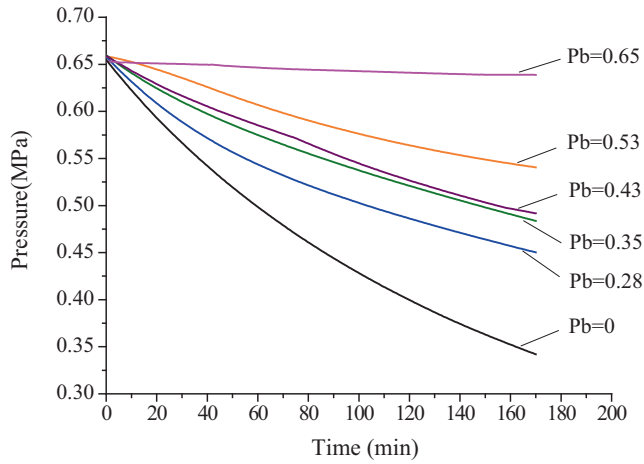


Fig. 3. Static leakage behaviors at different buffer pressures.

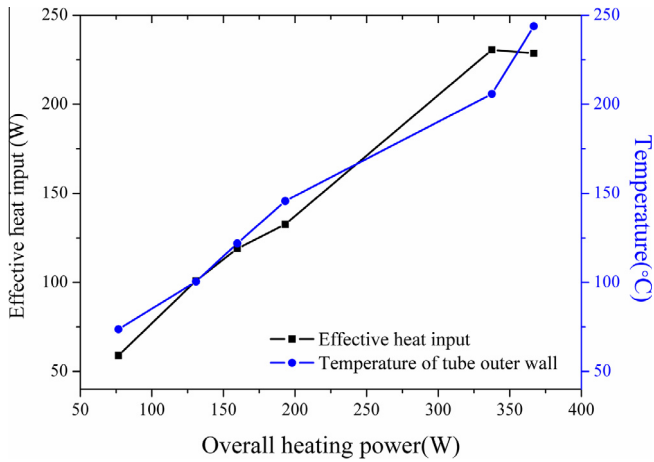


Fig. 4. Effective heat input and average wall temperature of the heater tubes vs. the overall heating power.

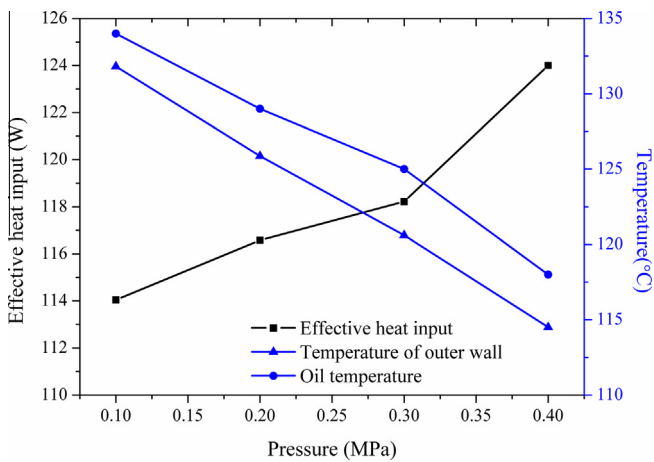


Fig. 5. Effective heat input and average temperatures at different gas pressures.

are made that the flow field is one-dimensional and the velocity in cylinder is in accordance with the displacer movement. The working gas cross-sectional mean velocity in the heater is written by [6],

$$u = (d_p/d_i)^2 \times (2x_{max}Nr/n \times 60) \quad (10)$$

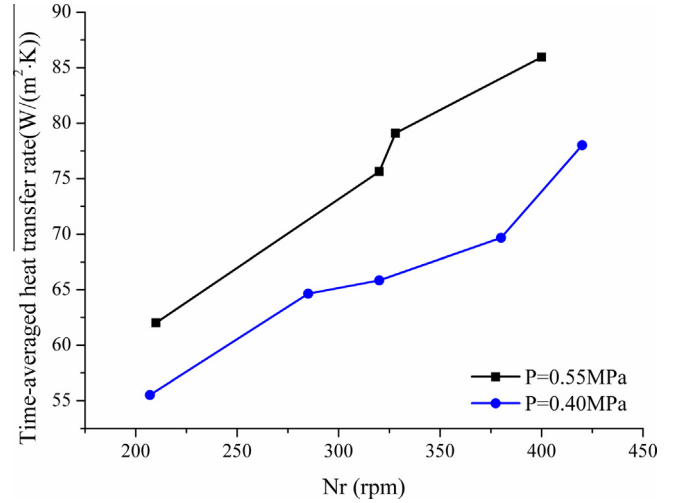


Fig. 6. Time-averaged heat transfer rates under different revolutions per minute.

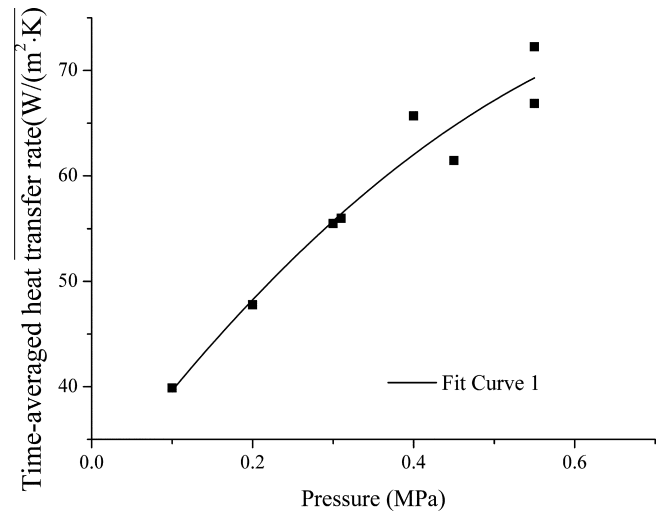


Fig. 7. Time-averaged heat transfer rates under different mean pressure values.

Thus the Reynolds number based on the cross-sectional mean velocity and the kinetic Reynolds number for the flow in the heater could be obtained as:

$$Re = ud_i/\nu \quad (11)$$

$$Re_\omega = \omega d_i^2/\nu \quad (12)$$

During the tests, the Re is in the range of 740–4110, and the corresponding Re_ω is 12–71. Taking oscillating effects into consideration, the heat transfer correlation in the heater could be expressed as follows. The correlation based on experimental results is regression fitted with the determination coefficient of $R^2 = 0.97$, seen in Fig. 8.

$$Nu = 0.0162Re_\omega^{0.40}A_0^{0.85} \quad (13)$$

where, $A_0 = x_{max}/d_i$ is a dimensionless fluid displacement.

3.2. Comparisons between experimental and theoretical results

Ratios of the experimental Nusselt number and calculated value by some theoretical correlations in Table 1 are plotted in Fig. 9. If considering the deviation of $\pm 20\%$ is acceptable, the calculated results from the Eqs. (c) and (i) are reasonable. It is noticed the Eq. (c)

Table 4
Cycle-averaged experimental data and uncertainty analysis.

Items	ω (rad/s)	Re_ω	A_0	UI (W)	ϕ_e (W)	w_{ϕ_e} (W)	$T_{w,in}$ (°C)	T_g (°C)	h_g (W/(m ² K))	Uncertainty (%)	Nu
1	21.36	36.18	181.05	130.9	100.96	7.7	100.45	79.82	55.00	7.63	5.54
2	21.68	33.91	181.05	159.6	118.97	8.4	121.87	96.24	52.15	7.06	5.07
3	22.51	26.69	181.05	337.5	230.63	12.5	205.70	160.31	57.10	5.42	4.89
4	33.51	56.04	181.05	130.9	100.14	7.7	99.14	82.41	67.26	7.69	6.73
5	21.68	33.91	181.05	161.5	126.63	8.5	121.87	96.24	55.51	6.72	5.39
6	29.85	47.86	181.05	161.5	125.14	8.5	112.81	91.05	64.64	6.80	6.35
7	33.51	53.18	181.05	161.5	124.28	8.5	114.44	93.23	65.84	6.84	6.44
8	39.79	62.68	181.05	161.5	123.51	8.5	114.70	94.78	69.67	6.89	6.79
9	43.98	71.28	181.05	161.5	128.60	8.5	107.37	88.85	78.01	6.61	7.70
10	33.51	12.93	181.05	161.5	114.04	8.5	131.76	99.62	39.87	7.46	3.85
11	33.51	25.99	181.05	161.5	116.58	8.5	125.81	98.38	47.77	7.29	4.62
12	33.51	39.28	181.05	161.5	118.22	8.5	120.55	96.61	55.48	7.19	5.39
13	33.51	53.18	181.05	161.5	124.00	8.5	114.44	93.23	65.69	6.86	6.42

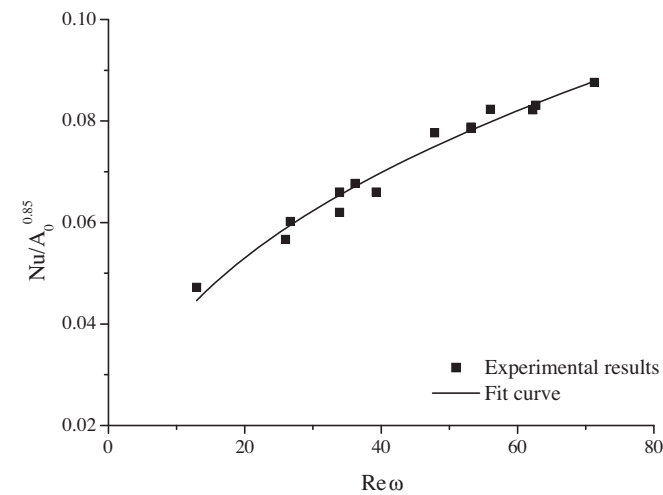


Fig. 8. Experimental Nusselt number as a function of Re_ω and A_0 .

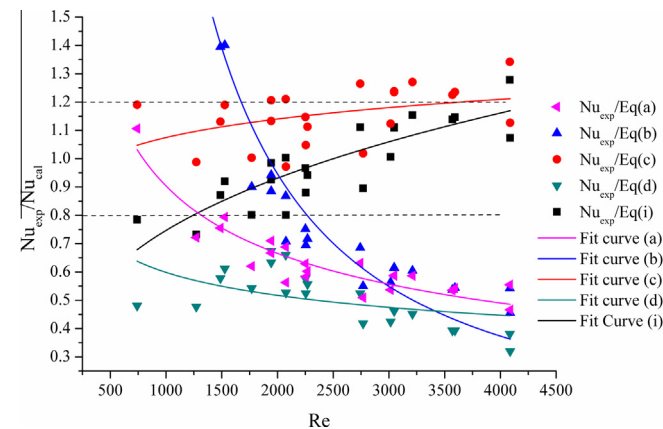


Fig. 9. Ratios of experimental and calculated Nu vs Re .

gives Nu values generally smaller than the experimental ones. The Eq. (i) over-predicts Nu values a little at low Re s, while under-predicts at high Re s. The other three equations, unidirectional turbulent correlations, give much bigger Nu for most Re s, and deviations become larger when Re increases, which was also noticed by Kuosa's [3].

These experiments confirm that the heat transfer characteristics of the oscillating flows are very different from those of the unidirectional steady flows, especially when Re is high. The kinetic

Reynolds number, the Womersley number and the dimensionless amplitude are important parameters for the oscillating flows.

4. Conclusions

A 36-tube Stirling engine heater was used to investigate the heat transfer of oscillating flow characteristics. The structure of the experimental apparatus and the testing conditions are similar to the operating conditions of real Stirling engines. The experimental results show that to increase the working gas pressure is a positive way to improve the effective heat input and to decrease the wall temperature of heater tubes, reducing the requirement on materials. The time-averaged heat transfer coefficients between the inner tube wall and the working gas increase with the working gas mass flow rate which is relative to the pressure and the oscillating frequency. An empirical oscillating heat transfer correlation is derived based on the kinetic Reynolds number and the dimensionless amplitude, i.e. $Nu = 0.0162Re_\omega^{0.40}A_0^{0.85}$, where Re and Re_ω are in the ranges of 740–4110 and 12–71, respectively. The estimated uncertainty for the heat transfer coefficients is usually within 7.69%, and the determination coefficient of a regression fitted correlation is 0.97. The experimental results are also compared with formerly proposed correlations, indicating that the classical unidirectional steady correlations are not suitable for oscillating flows, especially at high Re conditions. The empirical oscillating heat transfer correlation provides time-averaged heat transfer coefficients of oscillating flows, expecting to be valuable for practical design of tubular heaters.

Acknowledgements

The authors gratefully acknowledge the support from the National Natural Science Foundation of China (NO. 51276167), the National Science Foundation for Distinguished Young Scholars of China (No. 51125025), the Important Science & Technology Specific Projects of Zhejiang Province (No. 2012C01022-1), and the Zhejiang Provincial Natural Science Foundation of China (NO. LY12E06005).

References

- [1] T.W. Simon, J.R. Seume, A Survey of Oscillating Flow in Stirling Engine Heat Exchangers, University of Minnesota, Minnesota, 1988, pp. 71–75.
- [2] I. Urieli, D.M. Berchowitz, Stirling Cycle Engine Analysis, A. Hilger, Bristol, 1984, pp. 160–167.
- [3] M. Kuosa, K. Saari, A. Kankkunen, T.M. Tveit, Oscillating flow in a Stirling engine heat exchanger, Appl. Therm. Eng. 45 (2012) 15–23.
- [4] H.T. Shin, S. Nishio, Oscillation-controlled heat-transport tube (heat transfer coefficient in tubes in heating and cooling regions), Heat Transfer-Jpn. Res. 27 (6) (1998) 415–430.

- [5] A. Fan, D. Fulmer, J. Hartenstine, Experimental study of oscillating flow heat transfer, in: *Proceedings of MNHT2008 Micro/Nanoscale Heat Transfer International Conference*, Taiwan, 2008.
- [6] M. Kanzaka, M. Iwabuchi, Study on heat transfer of heat exchangers in the Stirling engine-heat transfer in a heated tube under the periodically reversing flow condition, *JSME Int. J.* 35 (1992) 641–646.
- [7] M. Kanzaka, M. Iwabuchi, Study on heat transfer of heat exchangers in the Stirling engine – performance of heat exchangers in the test Stirling engine, *JSME Int. J.* 35 (1992) 647–652.
- [8] P. Bouvier, P. Stouffs, J.P. Bardon, Experimental study of heat transfer in oscillating flow, *Int. J. Heat Mass Transfer* 48 (12) (2005) 2473–2482.
- [9] U. Akdag, A.F. Ozguc, Experimental investigation of heat transfer in oscillating annular flow, *Int. J. Heat Mass Transfer* 52 (11) (2009) 2667–2672.
- [10] S.M. Yang, W.Q. Tao, *Heat Transfer*, fourth ed., Higher Education Press, Beijing, 2006. pp. 246–270.
- [11] X.G. Tang, P. Cheng, Correlations of the cycle-averaged Nusselt number in a periodically reversing pipe flow, *Int. Commun. Heat Mass Transfer* 20 (2) (1993) 161–172.
- [12] F. Monte, Thermal analysis of the heat exchangers and regenerator in Stirling cycle machines, *J. Propul. Power* 13 (3) (1997) 404–411.
- [13] T. Zhao, P. Cheng, Oscillatory heat transfer in a pipe subjected to a laminar reciprocating flow, *ASME J. Heat Transfer* 118 (1996) 592–598.
- [14] C. Walther, H.D. Kühl, T. Pfeffer, S. Schulz, Influence of developing flow on the heat transfer in laminar oscillating pipe flow, *Forsch. Ingenieurwes.* 64 (3) (1998) 55–63.
- [15] J.P. Holman, *Experimental Methods for Engineers*, eighth ed., McGraw-Hill, New York, 2012. pp. 60–74.